

A COMPARISON OF FLEXIBLE AND RIGID RING BAFFLES FOR SLOSH SUPPRESSION

by

Luis R. Garza

Technical Report No. 1

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Prepared for

George C. Marshall Space Flight Center
National Aeronautics and Space Administration
Huntsville, Alabama

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Department of Mechanical Sciences

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APPROVED:



H. Norman Abramson, Director
Department of Mechanical Sciences

ABSTRACT

A test program to determine the effectiveness of flexible ring baffles as liquid slosh suppressors was conducted. A comparison of flexible and rigid ring baffles is presented in terms of liquid damping, first mode sloshing resonant frequency, and maximum baffle depth for no rotational slosh.

AUTHOR

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INTRODUCTION

The stability of liquid fuel filled launch vehicles depends largely on the suppression of propellant sloshing, particularly the first anti-symmetrical liquid mode for which the resultant sloshing force and overturning moments are the largest.

The sloshing problem has been thoroughly investigated. Many types of baffles have been analytically studied and experimentally examined to evaluate their effectiveness. The selection of the proper baffle depends largely on the liquid resonant frequency as in many vehicles the liquid resonant frequency must be carefully controlled to prevent dynamic coupling with the various vehicle structural components. In such a case the damping system employed must serve the dual purpose of damping and altering the liquid resonant frequency. Tank compartmentation or other means of raising the liquid natural frequency may be employed in such cases. Typical results for liquid sloshing in compartmented tanks have been presented in Ref. (1). When the liquid resonant frequency is not critical, ring baffles provide good damping with a minimum of added weight to the vehicle [Refs. (2, 3, 4 and 5) present a fairly complete study of rigid ring baffle damping]. Although the ring baffle damping-to-weight ratio is considered acceptable, the total weight of the baffle represents a fairly large amount of dead weight in launch vehicles with large tanks.

Lightweight flexible baffles have been suggested to decrease the baffle weight. D. G. Stephens, NASA Langley Research Center,

presented data in Ref. (6) on damping obtained for various flexible baffles in a rectangular tank. The relative damping (ratio of flexible baffle damping to rigid baffle damping) obtained by Stephens indicated that flexible baffles would not only reduce the baffle weight but would also increase the liquid damping effectiveness. These interesting results lead to the experimental studies for flexible ring baffles presented in this paper. Whereas the log decrement damping reported by Stephens were obtained from decay records, the data represented herein is from force response curves obtained from a force excited, liquid filled, cylindrical tank; furthermore, most of the data given here is for baffles near the free surface.

TANK CONFIGURATION AND TEST PROCEDURE

The experimental equipment and procedures utilized in the present work are similar to those employed in Ref. (7). For all tests, the following data were recorded: total resultant liquid force on the tank wall, tank excitation amplitude, and liquid slosh height. All the tests were carried out in a 17.75 inch diameter tank. The tank was made in two halves, each 17.75 inches long and the baffle was clamped between the flanges joining the tank halves, as shown in Figure 1. With this configuration, the resulting damping and changes in natural frequencies are fully attributed to the baffle. For all these tests the bare wall tank (no baffle) damping was neglected as it is very small, at the translation excitation amplitudes tested, and the liquid goes into rotation at the first antisymmetric resonant frequency.

The baffles examined were similar to those of Ref. (4), with a width to radius ratio $W/R = 0.157$. The rigid baffle was made of 1/8 inch thick 2024-T6 aluminum. The flexible baffles were made of Mylar of the following thicknesses: 0.002", 0.003", 0.005" and 0.0075". All tests were conducted at three excitational amplitudes $X_0/d = 0.00152, 0.0035, 0.0067$, and baffle depths ranging from $d_s/R = 0$, to a baffle submergence of $d_s/R = 0.4$. The damping values in all cases were computed from the force response curve by the half-band-width technique.

LIQUID DAMPING RIGID AND FLEXIBLE BAFFLES

A. Rigid Baffle Damping

The first part of the test program was directed toward comparing the damping values obtained in the present 17.75 inch diameter tank to those values obtained in the 14.4 inch diameter tank of Ref. (4). The comparison of these two different tests for three excitational amplitudes is presented in Figure 2, from which it can be seen that the results from the two tanks are in good agreement. It should be noted that for the present tank no data was obtained at baffle depths greater than $d_s/R = 0.2$ at the large excitation amplitude $X_o/d = 0.0067$ because at the resonant frequency the liquid went into a rotational type of sloshing. This did not occur in the previous tests of Ref. (4).

B. Flexible versus Rigid Baffles

A comparison of the damping ratio versus baffle depth for the rigid and the various flexible baffles for a translation amplitude $X_o/d = 0.00152$ is presented in Figure 3. At this value of excitation amplitude the damping from the rigid baffle is slightly greater at baffle depths of $d_s/R = 0$ to approximately 0.025; however, for greater baffle depths the various flexible baffles are seen to be more effective.

A comparison of the relative damping (ratio of flexible baffle

damping to rigid baffle damping) for this translation amplitude, is presented in Figure 4; the 0.002" and 0.003" Mylar baffles appear to be the most effective ones. No rotational slosh was encountered in any of these tests; however, it is believed that for greater baffle depths, $d_s/R > 0.4$, rotational slosh would have occurred.

Figure 5 presents damping ratio versus baffle depth for the various baffles, at a translation amplitude $X_o/d = 0.0035$. It can be noted that again the rigid baffle is more effective near the free surface, but the flexible baffles become more effective at baffle depths greater than $X_o/d > 0.1$. Also shown in this figure are the baffle depths for which rotational slosh was encountered for the various baffles. The 0.003" Mylar baffle was tested to a baffle depth $d_s/R = 0.4$ without encountering rotation; however, rotation was encountered with just a slight increase in baffle depth. The relative damping, at this same excitation amplitude, is presented in Figure 6 from which it can be seen that the flexible baffles offer approximately 15 to 20 percent more damping than do the rigid baffles. No distinct advantage can be seen from any one flexible baffle to another, at this excitation amplitude, other than that the 0.003" Mylar baffle suppressed rotational sloshing to a greater depth.

Figure 7 presents damping ratio versus baffle depth for the various baffles at a translation amplitude $X_o/d = 0.0067$. The results near the free surface are similar to those for the smaller excitation

amplitudes. For all baffles, the damping produced at baffle depths $d_s/R = 0.05$ to 0.1 was so great that the damping coefficient could not be obtained from the force response with the half-band-width technique. For this excitation amplitude the liquid went into rotation at baffle depths $d_s/R = 0.25$ for the rigid and various flexible baffles, and at $d_s/R = 0.30$ for the $0.002''$ thick Mylar baffle. The relative damping for this excitation amplitude is presented in Figure 8. It may be noted that at baffle depths $d_s/R = 0.2$ the flexible baffles are slightly better than the rigid baffles; however, at greater depths no damping data for these baffles could be obtained because of the rotational slosh.

It is questionable whether the flexible baffles would offer more damping than the rigid baffles for large excitation amplitudes; however, it is believed that their damping would still be at least comparable to that of a rigid baffle.

An approach to the selection of the appropriate flexible baffle for a given tank diameter is presented in the Appendix. For this particular tank, the $0.003''$ Mylar baffle more nearly corresponds to the recommendations of the Appendix. Figure 9 shows the results obtained at a fairly small translation excitation amplitude for the rigid baffle and the $0.003''$ Mylar baffle. For the rigid baffle no damping values were computed at baffle depths less than $d_s/R = 0.05$ since the damping was out of the range of the half-band-width technique. At

baffle depths greater than 0.05 the flexible baffle damping is approximately twice that of the rigid baffle.

A comparison of baffle width effect is presented in Figure 10. It can be seen that the damping for the $W/R = 0.157$ baffle is much greater than for the $W/R = 0.10$ baffle. It should also be noted that rotational slosh was encountered at shallower depths for the $W/R = 0.10$ baffle, particularly for the comparison of the two flexible baffles. For both baffle widths, the flexible baffle damping is slightly greater than for a rigid baffle.

C. Effect of Period and Flexibility Parameters

Stephens (Ref. 6) correlated his data by three dimensionless parameters. The first was a period parameter, $P = UT/W$, where U is the liquid velocity past the baffle, T is the period of the sloshing oscillation, and W is the baffle width; this parameter also is equal to the slosh height times the exponential function that accounts for the baffle depth (Ref. 3) divided by the baffle width. The second parameter was a flexibility parameter, $F = (W/t)^3 (1 - \mu^2) \left(\frac{\rho W^2}{ET^2} \right)$ where t is the baffle thickness, μ is Poisson's ratio, ρ is the liquid density, and E is the modulus of elasticity. The third parameter was the relative damping $\frac{\gamma_{flex}}{\gamma_{rigid}}$, where γ_{flex} is the damping ratio as provided by a flexible baffle and γ_{rigid} is the damping ratio provided by a rigid baffle of the same width under similar conditions.

No period parameter could be determined for shallow baffle depths because of difficulties in defining and measuring the small slosh heights then encountered. Also, for these depths the sloshing mode for a ring baffled cylindrical tank appears to be a combination of antisymmetric and symmetric sloshing. For baffle depths $d_s/R = 0.15$ and greater, the first antisymmetric slosh becomes predominant and the period parameter can be computed from the measurement of the slosh height. For the various flexible baffles tested, four values of the flexibility parameters could be computed.

The data points were first plotted for each Mylar baffle and a geometrically similar rigid baffle, as a function of damping ratio versus period parameter. Figures 11, 12, 13 and 14 present these various plots. For each baffle depth the damping ratio appears to decrease almost linearly with a decrease in period parameter. Also, the damping ratio decreases and the period parameter increases with an increase of baffle depth, for both the flexible and the rigid baffles. It can also be noted that, for an equivalent excitation amplitude, the period parameter for the rigid baffle is considerably greater than that for a flexible baffle since the slosh heights were greater for the rigid baffle.

To compare the relative damping of the flexible and rigid baffles against a flexibility parameter, three period parameters were selected and the relative damping determined from the damping ratio

versus period parameter plots. Figures 15, 16, 17 and 18 present the relative damping versus flexibility parameter for four baffle depths. It can be seen from these four figures that for a period parameter range $P = 1.6$ to 3.0 , the flexible to rigid baffle relative damping appears to be between 40 and 50 percent higher, for all values of baffle flexibility parameter.

LIQUID RESONANT FREQUENCIES

The first sloshing mode frequency for a ring baffled cylindrical tank is dependent on the baffle location below the liquid free surface and on the translation amplitude of the tank. Shown in Figures 19, 20 and 21 are the first mode liquid resonant frequencies in terms of the dimensionless parameter $\frac{\omega^2 d}{a}$, where ω^2 is the liquid resonant frequency squared, d is the tank diameter, and a is the axial acceleration versus the ring baffle depth d_s/R , for the three values of translation amplitude. From these figures it can be seen that the resonant frequency is increased above that of a bare wall tank when the baffle is very near the liquid free surface. As the baffle is submerged the resonant frequency decreases; however, the higher resonant frequency can be retained with an increase of tank translation amplitude. No first mode liquid resonant frequency was noted between baffle depths $d_s/R = 0.05$ and 0.1 for the largest translation amplitude $X_o/d = 0.0067$. For baffle depths close to the free surface (less than half the baffle width) the resonant frequency is well below that for a non-baffled tank. As the baffle depth is continually increased the liquid resonant frequency gradually increases and approaches that of a non-baffled tank. For the smallest excitation amplitude $X_o/d = 0.00152$ the lowest liquid resonant frequencies were recorded at the shallow depths; however, the bare wall liquid resonant frequencies were also

approached at much shallower baffle depths than for the larger excitation amplitudes. For all excitation amplitudes where the baffle is at the free surface, the most rigid baffle yielded the highest liquid resonant frequency. For baffle depths just under the free surface the lowest resonant frequency occurred for the more rigid baffles.

DISCUSSION

In general, test results indicate that flexible ring baffles provide damping equal to or better than similar rigid baffles. The relative damping, (ratio of flexible baffle damping to rigid baffle damping) varies from an approximate value of 2 for small tank amplitudes, $X_o/d = 0.00085$, to a value slightly greater than 1 for tank amplitudes $X_o/d = 0.0067$. For these series of tests, the more flexible baffles provided the higher damping values and also maintained a more constant liquid resonant frequency throughout the baffle depth range investigated.

Tests with a flexible (0.0012" thick) aluminum baffle were attempted for the greater tank amplitudes; however, tears occurred around the baffle inner edge and the tests were discontinued. Tearing of the baffle resulted from small initial nicks which occurred in cutting the inner radius of the baffle.

There is no question but that flexible baffles can reduce baffle weight by a considerable amount; however, the properties of the baffle material at cryogenic temperatures should be better known. It may also be necessary to investigate the stresses at the baffle inner edge to insure that tearing of the baffle does not occur.

ACKNOWLEDGMENTS

The author gratefully acknowledges the guidance and contributions of Drs. H. N. Abramson, Helmut F. Bauer, and particularly Franklin T. Dodge and Wen-Hwa Chu for the analysis of the flexible baffle resonant frequency. He also wishes to acknowledge the valuable assistance of Messrs. Dennis C. Scheidt and Richard F. Roemer in conducting many of the tests and assisting in the data reduction; Mr. David M. DeArmond for preparing all the illustrations, and Mrs. Adeline Raeke for typing this report.

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APPENDIX

It seems reasonable to assume that the highest damping, at least for large baffle submergence depths, will be given by a flexible baffle whose natural frequency (for the $\cos \theta$ mode) in liquid is the same as the sloshing natural frequency. (For maximum damping, the damping force should lag the excitation by 90° , and this is the phase angle between the excitation of an elastic structure and its motion exactly at resonance.)

The frequency of the baffle was computed from the elastic plate equation

$$D \nabla^4 y = q$$

where y is the baffle displacement, D is the flexural rigidity, and q is the loading. The loading is composed of the inertia of the baffle, $\rho_b h \ddot{y}$ (h is the baffle thickness and ρ_b the density of the baffle material), and the apparent inertia of the liquid, which is approximated here by the same inertia as that exerted on a rigid plate moving with the velocity of the flexible baffle at that point, $\rho W \ddot{y} [1 - (\frac{r}{W})^2]^{1/2}$. Thus, if $y = \bar{y}(r) \cos \theta \cos \Omega t$, $D \nabla^4 \bar{y} - \frac{\Omega^2}{\Omega_0^2} [1 + M_a \sqrt{1 - (\frac{r}{W})^2}] \bar{y} = 0$, where $\Omega_0^2 = D/\rho_b h W^4$ and $M_a = \rho W/\rho_b h$. This equation is put into finite difference form and solved by a matrix eigenvalue routine. The finite difference equation is

$$\{A\} \{\bar{y}\} - \left(\frac{\Omega}{\Omega_0}\right)^2 \{M\} \{\bar{y}\} = 0$$

Let the baffle width be broken up into N equidistant intervals of width Δ , i. e., $N\Delta = W$. Let the start of the first interval be at the tank wall-baffle intersection and call it $n=0$. Any other interval will then be located at $n\Delta$, $0 \leq n \leq N$.

Then, the elements of $\{M\}$ are:

$$M_{nn} = 1 + M_a [W^2 - (n\Delta)^2]^{1/2}$$

$$M_{n,\ell} = 0 \quad \text{if } n \neq \ell.$$

The elements $A_{n,\ell}$ of the matrix $\{A\}$ are defined in terms of the following parameters $C_{n,\ell}$

$$C_{n,n+2} = \frac{1}{\Delta^4} - \frac{1}{(R - n\Delta)\Delta^3}$$

$$C_{n,n+1} = -\frac{4}{\Delta^4} + \frac{2}{(R - n\Delta)\Delta^3} - \frac{1}{(R - n\Delta)^2\Delta^2} - \frac{3}{(R - n\Delta)^3\Delta}$$

$$C_{n,n} = \frac{6}{\Delta^4} + \frac{2}{(R - n\Delta)^2\Delta^2} - \frac{3}{(R - n\Delta)^4}$$

$$C_{n,n-1} = -\frac{4}{\Delta^4} - \frac{2}{(R - n\Delta)\Delta^3} - \frac{1}{(R - n\Delta)^2\Delta^2} + \frac{3}{(R - n\Delta)^3\Delta}$$

$$C_{n,n+2} = \frac{1}{\Delta^4} + \frac{1}{(R - n\Delta)\Delta^3}$$

Taking into account the boundary conditions on the baffle

(built-in at $r=0$, free at $r=W$) the elements of $\{A\}$ are as follows:

1. For $n = N$:

$$A_{N,N} = C_{N,N} + 2C_{N,N+2} \left\{ \left[\frac{K^2 + K + 3 - \nu}{K^2} \right] \left[\frac{2K^2 + \nu}{K(K - \nu)} \right] + \frac{3 - \nu - 2K^2}{K^3} \right\} \\ + \left\{ \frac{2K^2 + \nu}{K(K - \nu)} \right\} C_{N,N+1}, \quad \text{with } K = \frac{R - W}{\Delta}$$

$$A_{N,N-1} = C_{N,N-1} + 2C_{N,N+2} \left\{ \left[\frac{K^2 + K + 3 - \nu}{K^2} \right] \left[-\frac{K + \nu}{K - \nu} \right] \right. \\ \left. - \left[\frac{K^2 - K + 3 - \nu}{K^2} \right] \right\} + C_{N,N+1} \left\{ -\frac{K^2 + \nu}{K - \nu} \right\}$$

$$A_{N,N-2} = C_{N,N-2} + C_{N,N+2}$$

$$A_{N,N+1} = A_{N,N+2} = 0$$

All other $A_{N,\ell}$ are zero.

2. For $n = N-1$:

$$A_{N-1,N} = C_{N-1,N} + C_{N-1,N+1} \left[\frac{2K^2 + \nu}{K(K - \nu)} \right]$$

$$A_{N-1,N-1} = C_{N-1,N-1} + C_{N-1,N+1} \left[-\frac{K + \nu}{K - \nu} \right]$$

$$A_{N-1,N-2} = C_{N-1,N-2} \qquad A_{N-1,N-3} = C_{N-1,N-3}$$

All other $A_{N-1,\ell}$ are zero.

3. For $n = 0$:

All $A_{0,\ell}$ are zero.

4. For $n = 1$:

All $A_{1,\ell}$ are zero.

5. For $n = 2$:

$$A_{2,0} = 0$$

$$A_{2,1} = 0$$

$$A_{2,2} = C_{2,2}$$

$$A_{2,3} = C_{2,3}$$

$$A_{2,4} = C_{2,4}$$

All other $A_{2,\ell}$ are zero

6. For $3 \leq n \leq N-2$:

$$A_{n,n+2} = C_{n,n+2}$$

$$A_{n,n+1} = C_{n,n+1}$$

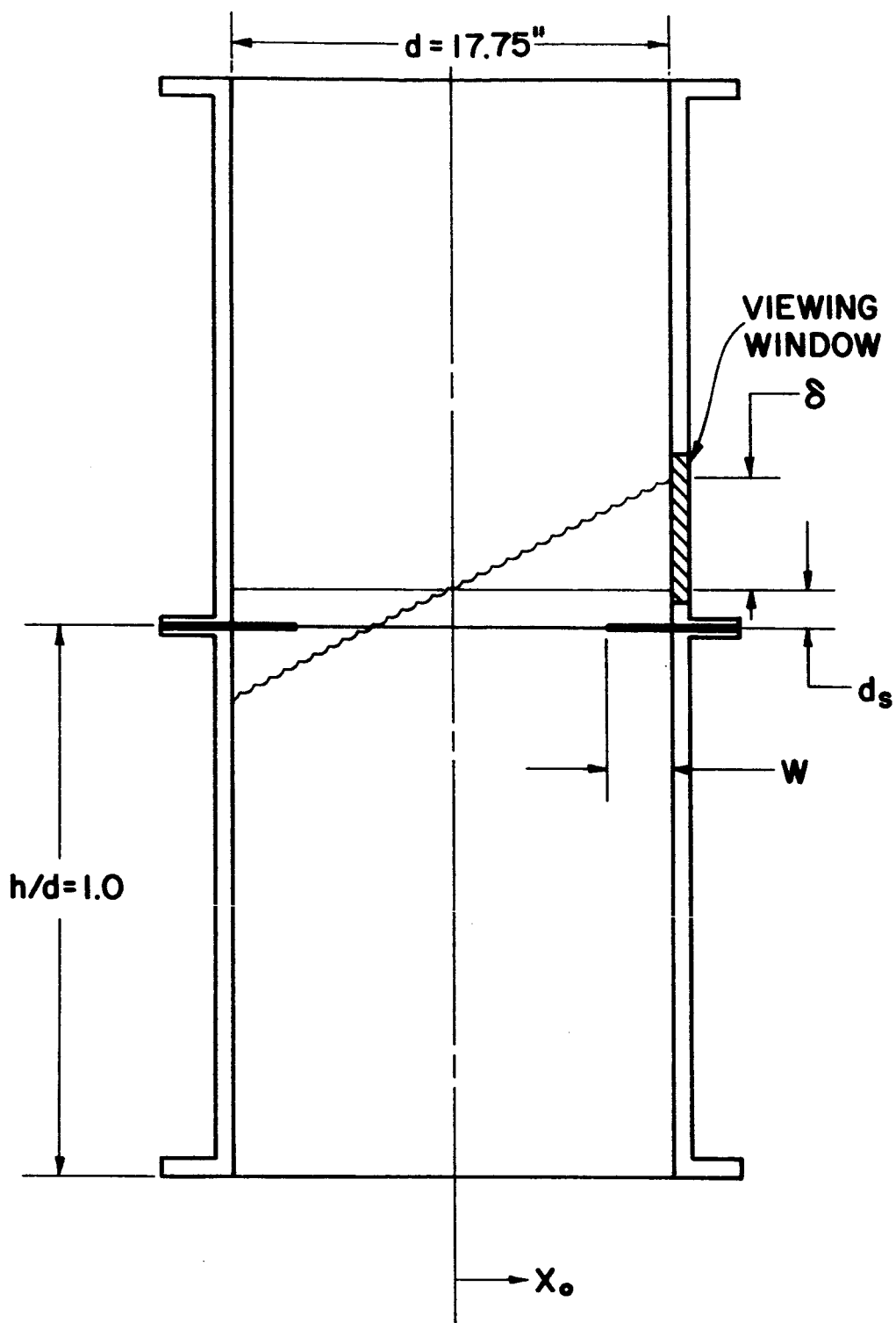
$$A_{n,n} = C_{n,n}$$

$$A_{n,n-1} = C_{n,n-1}$$

$$A_{n,n-2} = C_{n,n-2}$$

All other $A_{n,\ell}$ are zero.

Results of calculations using the previous frequency equation are shown in Figure 22. As can be seen, the 0.003" Mylar baffle has a frequency quite close to the sloshing frequency.



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FIGURE 1. TANK CONFIGURATION

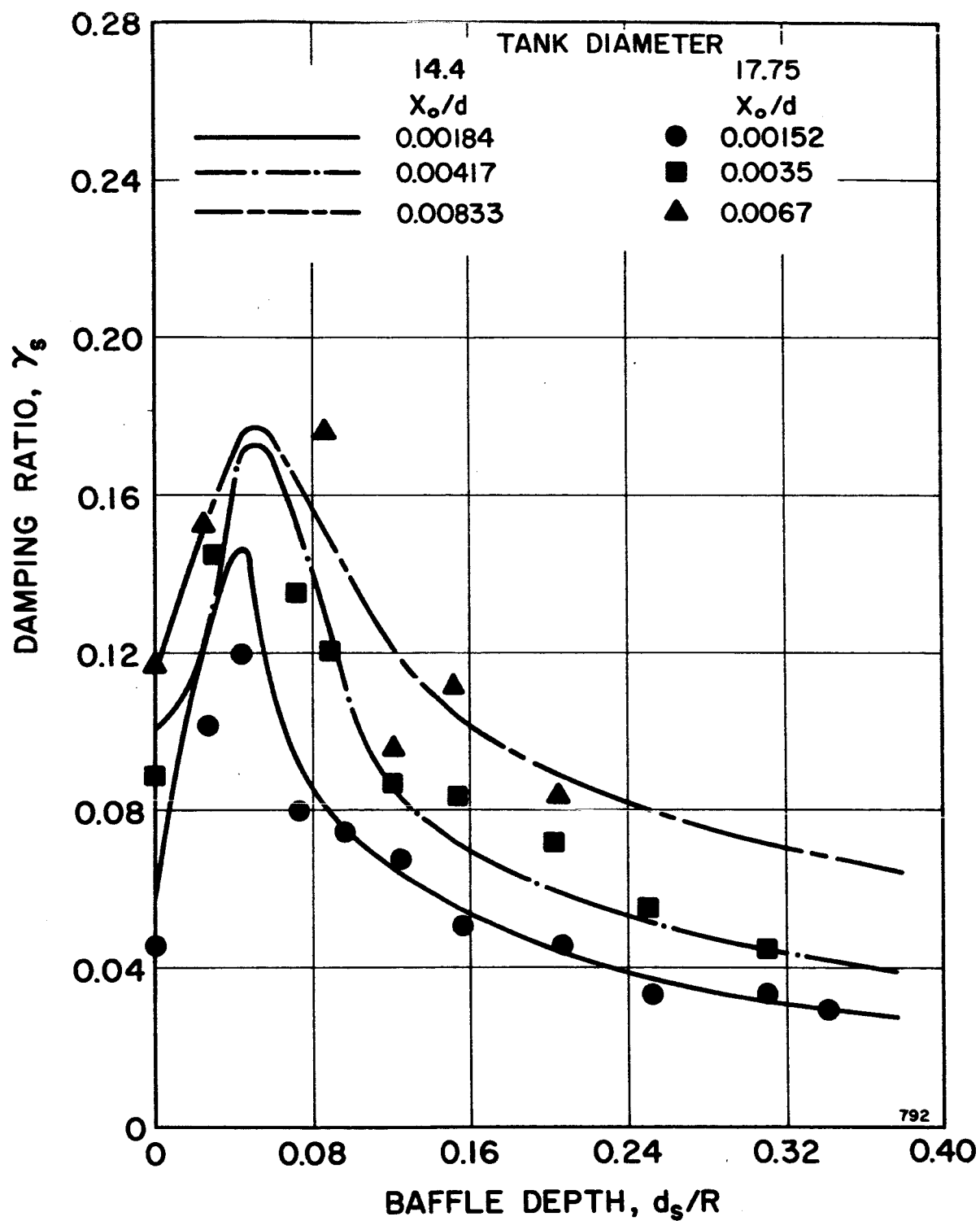


FIGURE 2. A COMPARISON OF RIGID BAFFLE LIQUID DAMPING FOR TWO DIFFERENT DIAMETER TANKS

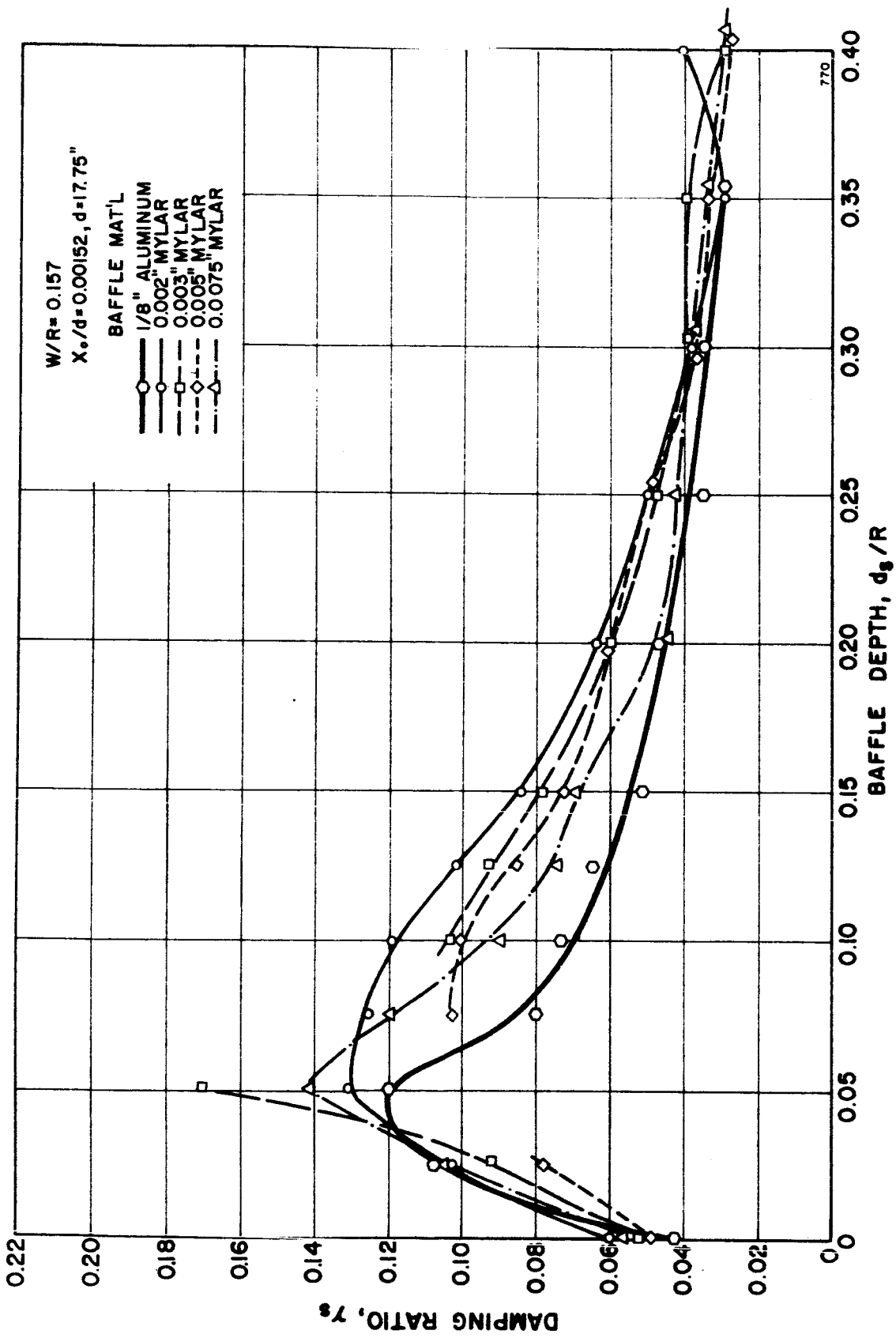


FIGURE 3. A COMPARISON OF RIGID AND FLEXIBLE BAFFLE LIQUID DAMPING
FOR A TANK AMPLITUDE $X_0/d = 0.00152$

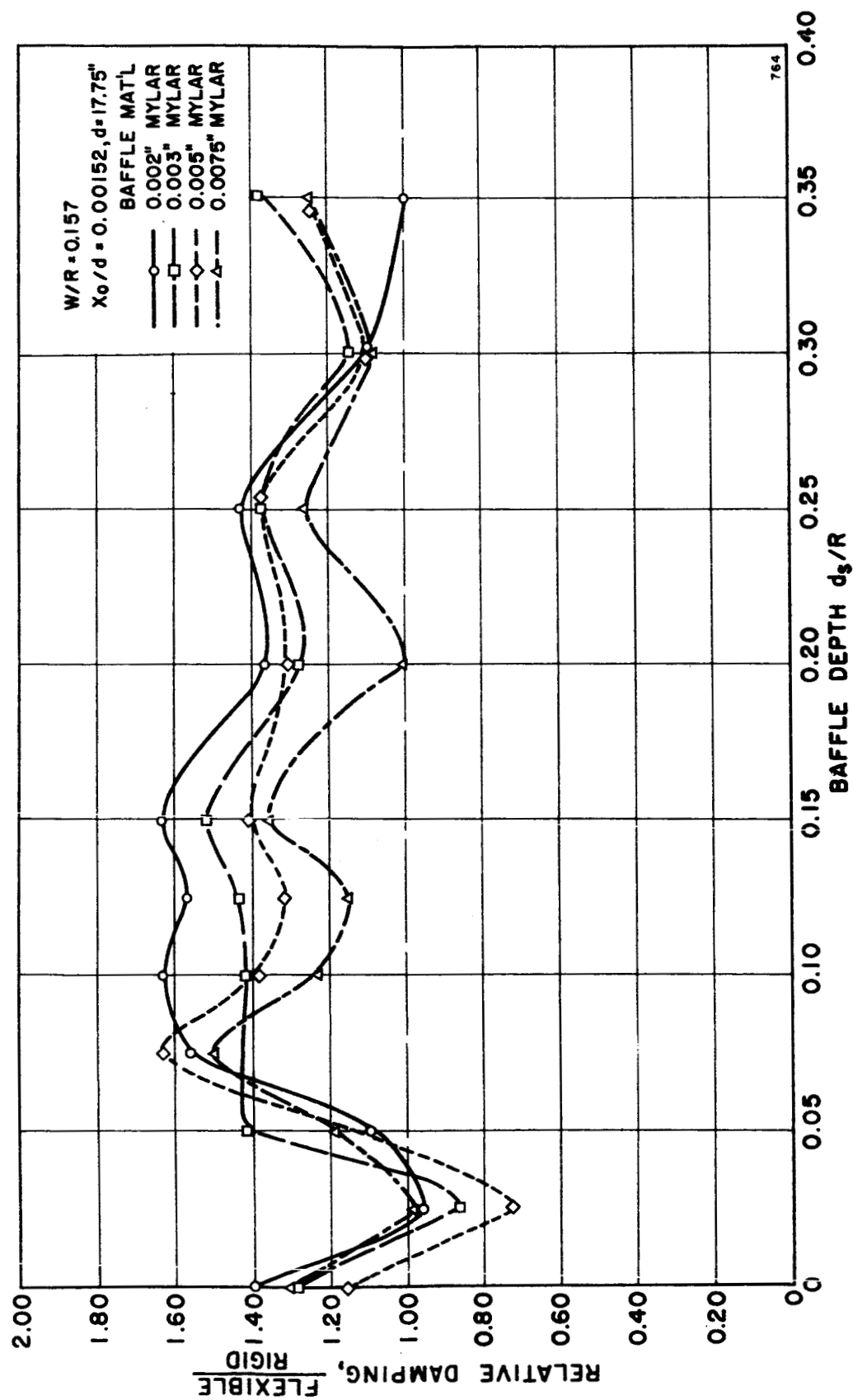


FIGURE 4. RELATIVE DAMPING OF VARIOUS FLEXIBLE BAFFLES FOR A
 TANK AMPLITUDE $X_0/d = 0.00152$

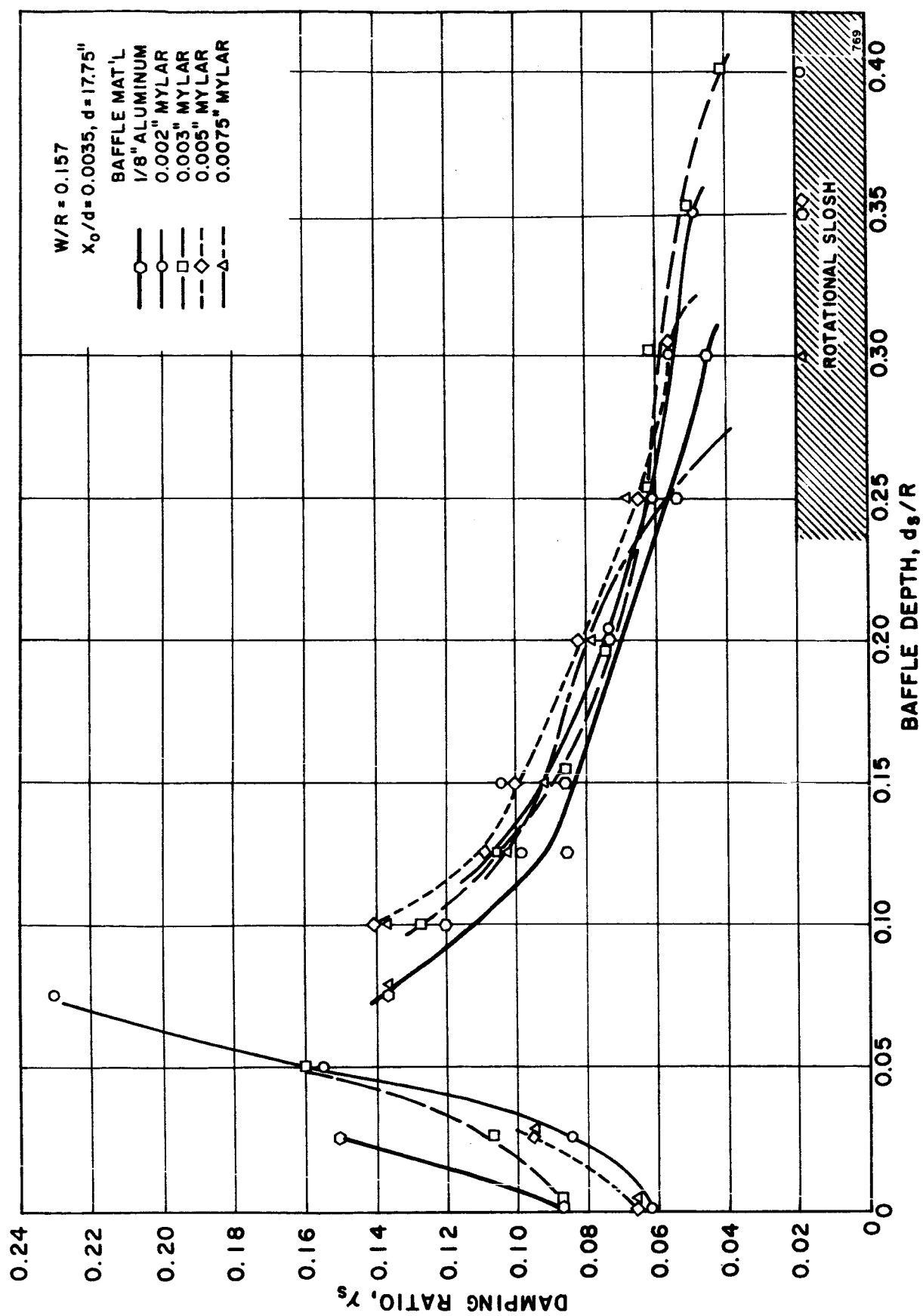


FIGURE 5. A COMPARISON OF RIGID AND FLEXIBLE BAFFLE LIQUID DAMPING
 FOR A TANK AMPLITUDE $X_0/d = 0.0035$

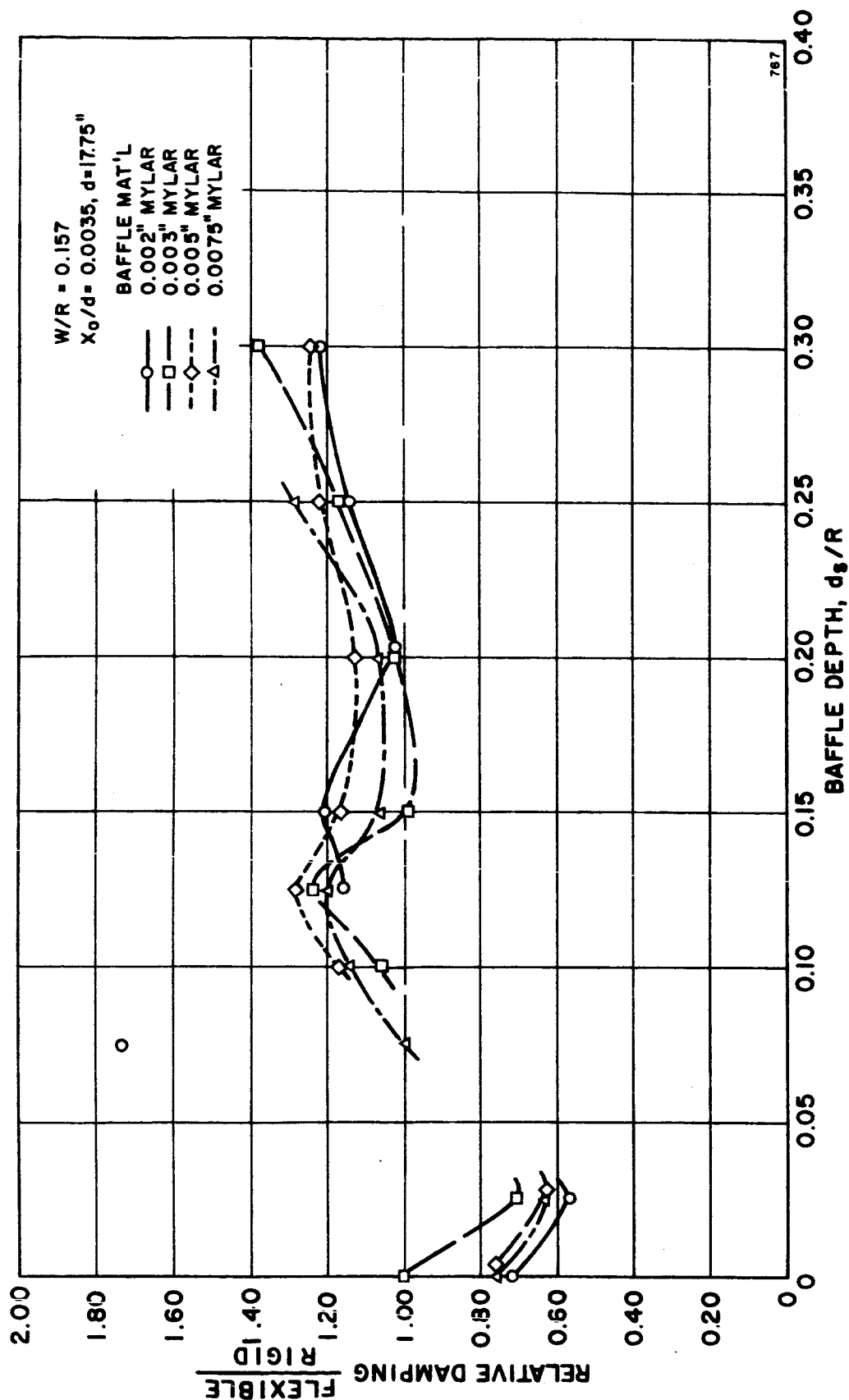


FIGURE 6. RELATIVE DAMPING OF VARIOUS FLEXIBLE BAFFLES FOR A
TANK AMPLITUDE $X_0/d = 0.0035$

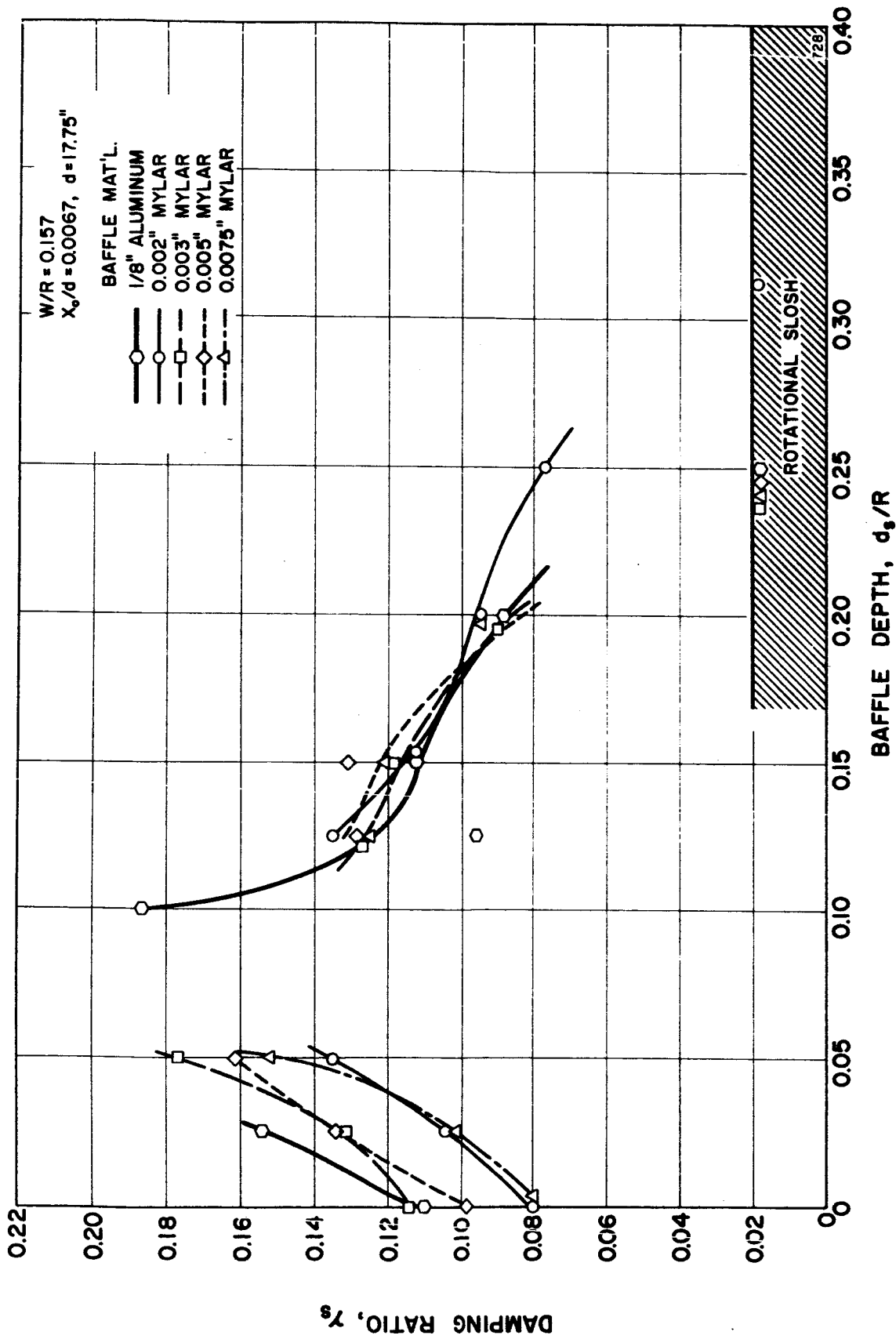


FIGURE 7. A COMPARISON OF RIGID AND FLEXIBLE BAFFLE LIQUID DAMPING
FOR A TANK AMPLITUDE $X_0/d = 0.0067$

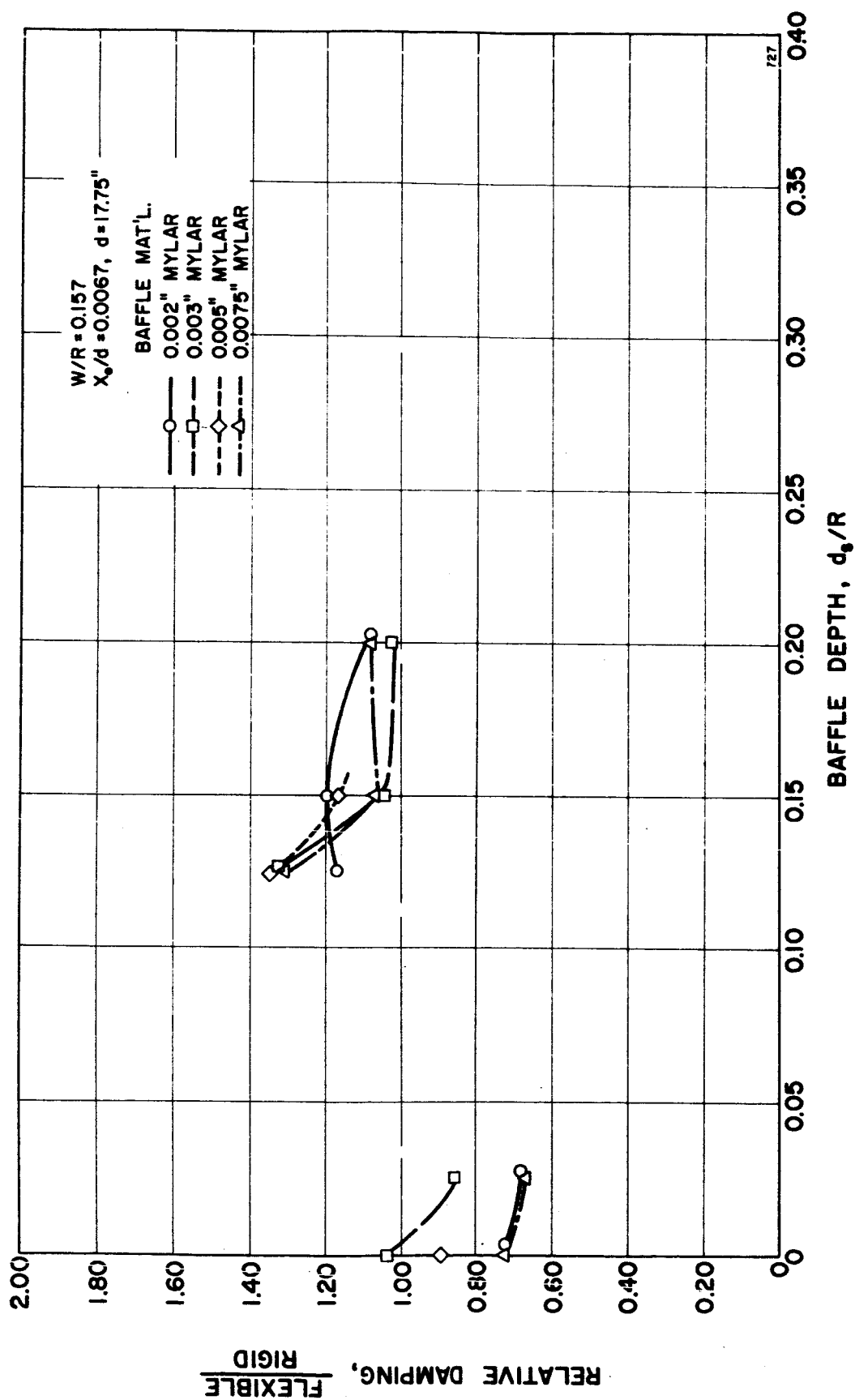


FIGURE 8. RELATIVE DAMPING OF VARIOUS FLEXIBLE BAFFLES FOR A
TANK AMPLITUDE $X_0/d = 0.0067$

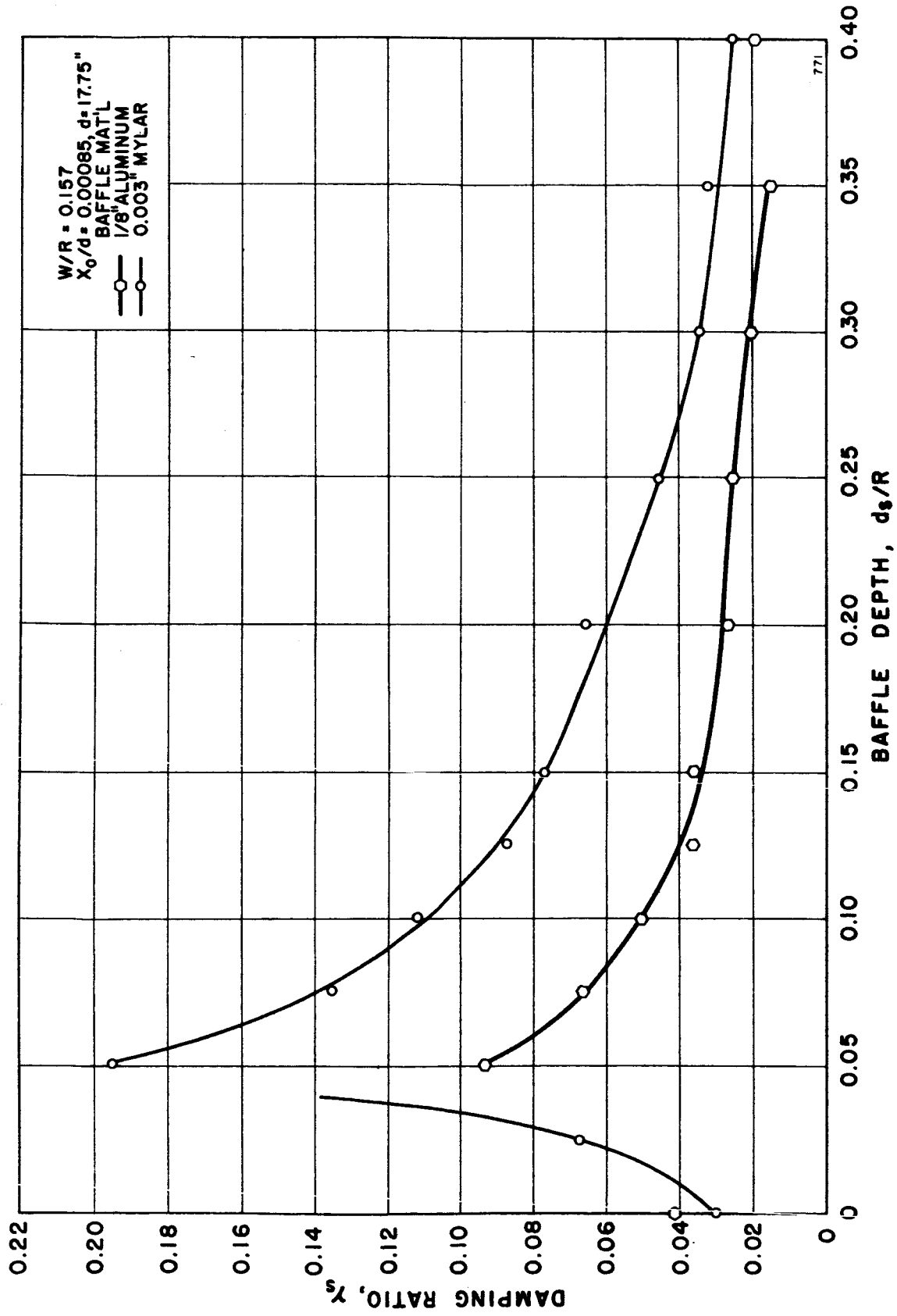


FIGURE 9. COMPARISON OF DAMPING RATIOS FOR A RIGID BAFFLE vs A 0.003" THICK MYLAR BAFFLE FOR A TANK AMPLITUDE $X_0/d = 0.00085$

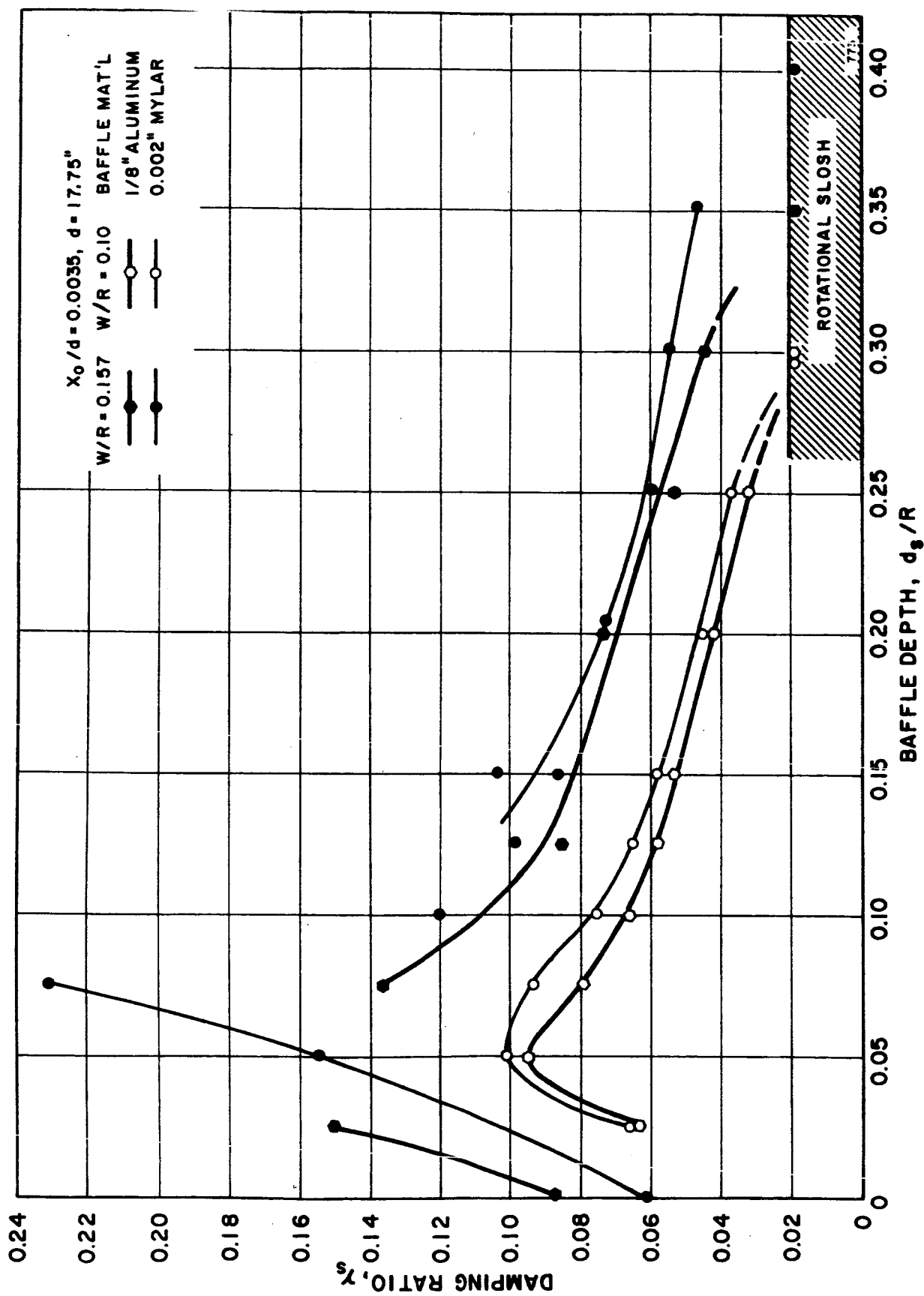


FIGURE 10. A COMPARISON OF BAFFLE WIDTH EFFECT ON LIQUID DAMPING FOR A TANK AMPLITUDE $X_0/d = 0.0035$

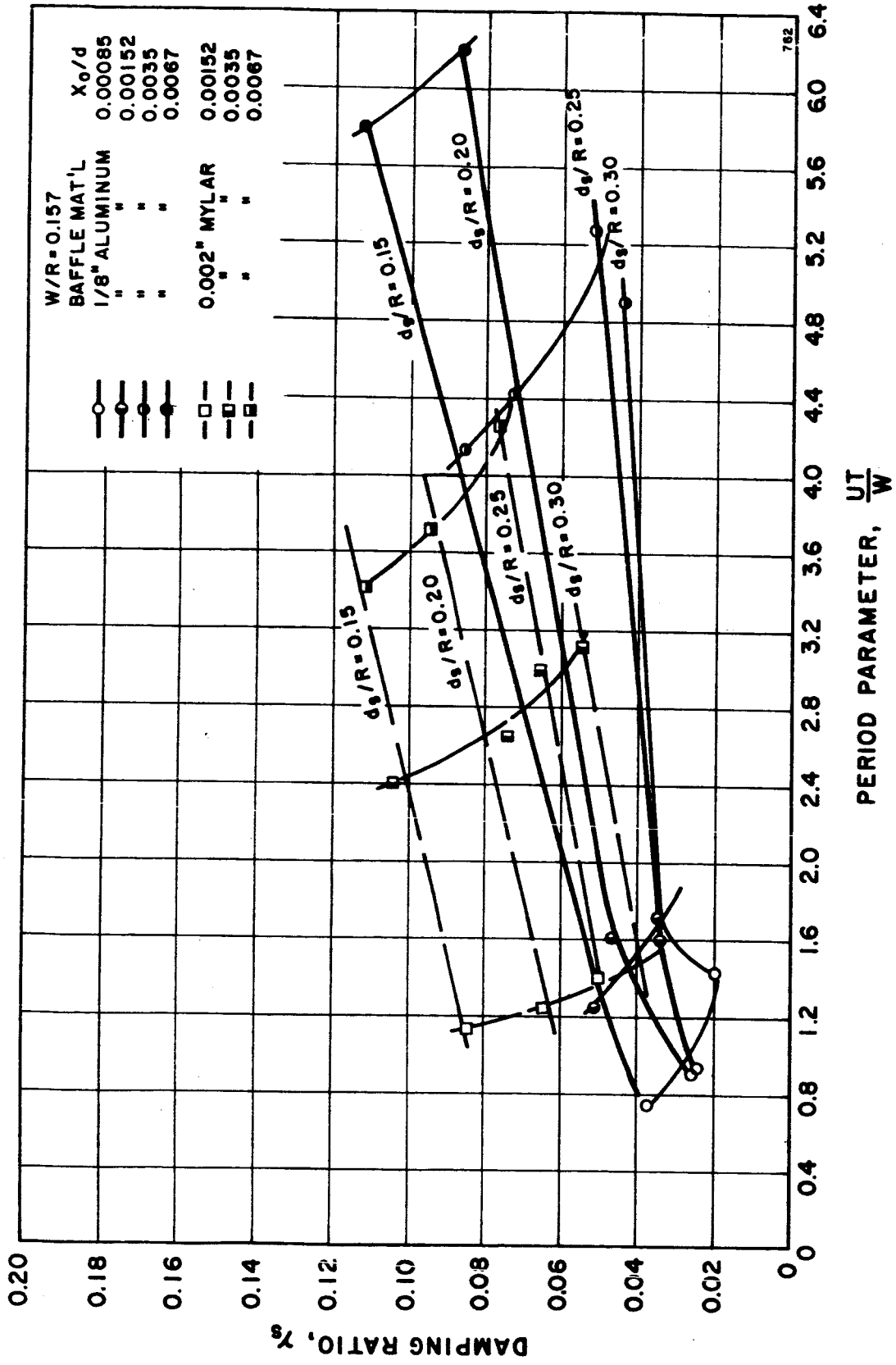


FIGURE 11. A COMPARISON DAMPING RATIO vs PERIOD PARAMETER FOR A RIGID AND A 0.002" THICK MYLAR BAFFLE

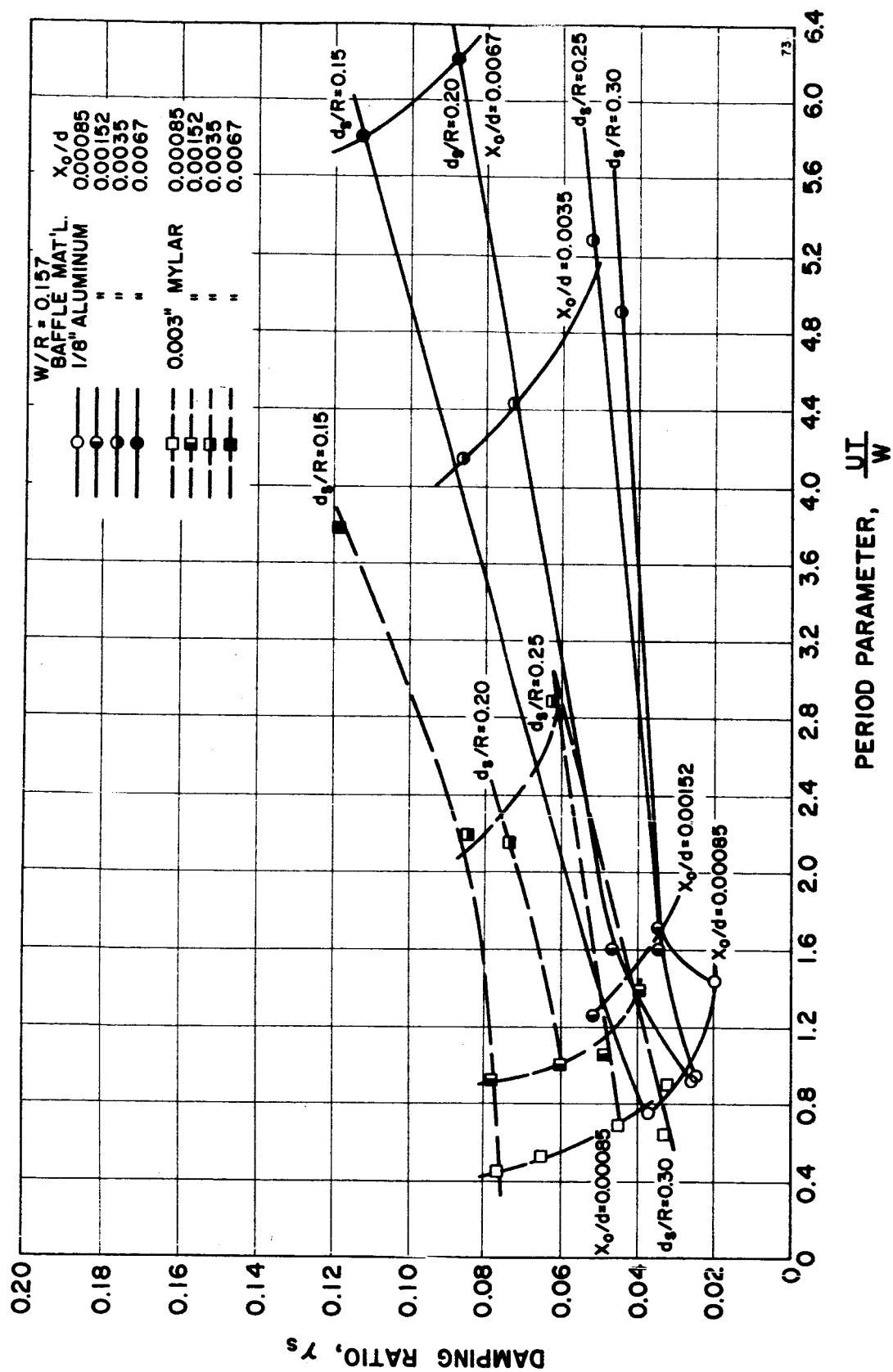


FIGURE 12. A COMPARISON DAMPING RATIO vs PERIOD PARAMETER FOR A RIGID AND A 0.003" THICK MYLAR BAFFLE

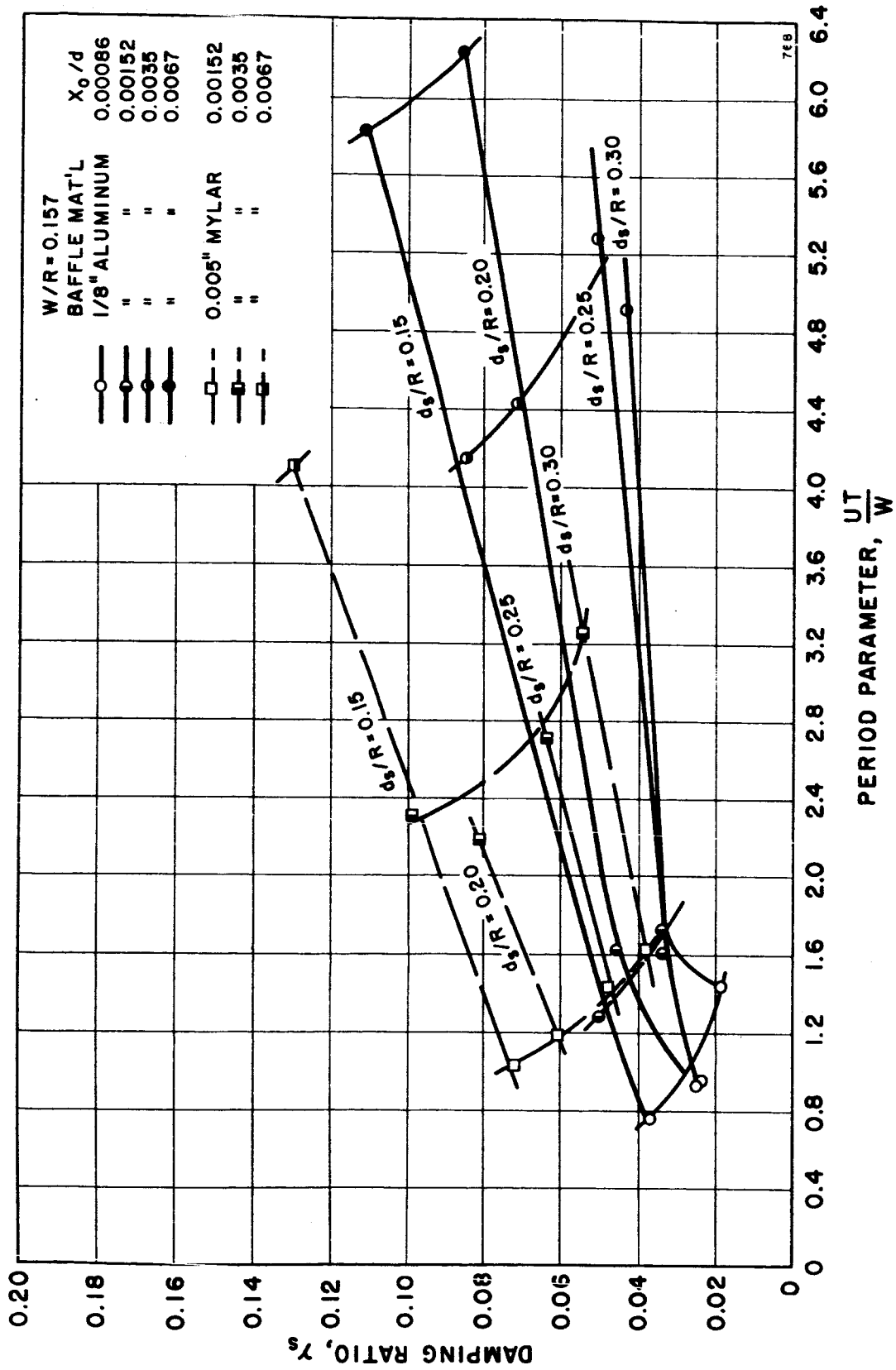


FIGURE 13. A COMPARISON DAMPING RATIO vs PERIOD PARAMETER FOR A RIGID AND A 0.005" THICK MYLAR BAFFLE

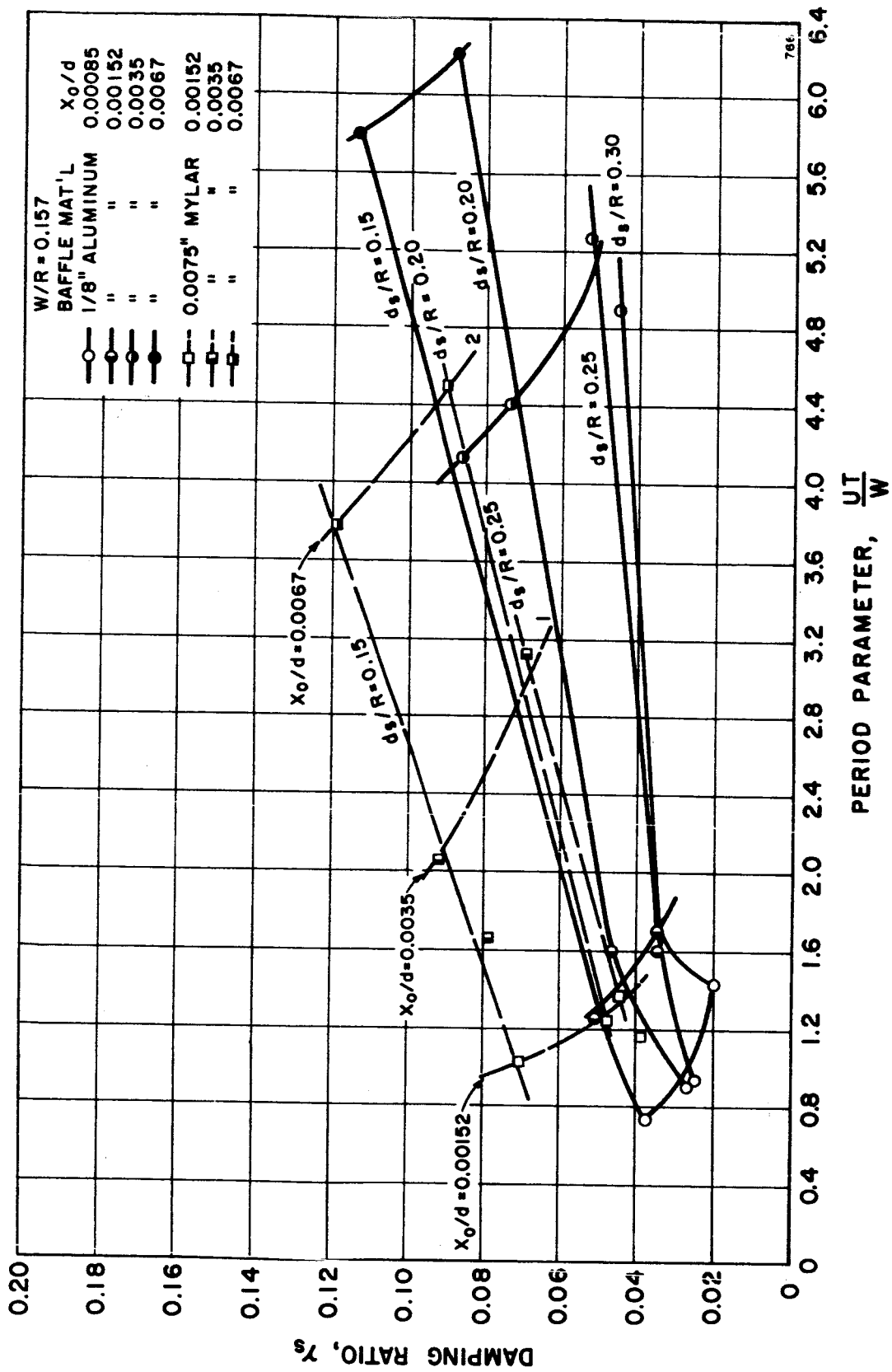


FIGURE 14. A COMPARISON DAMPING RATIO vs PERIOD PARAMETER FOR
A RIGID AND A 0.0075" THICK MYLAR BAFFLE

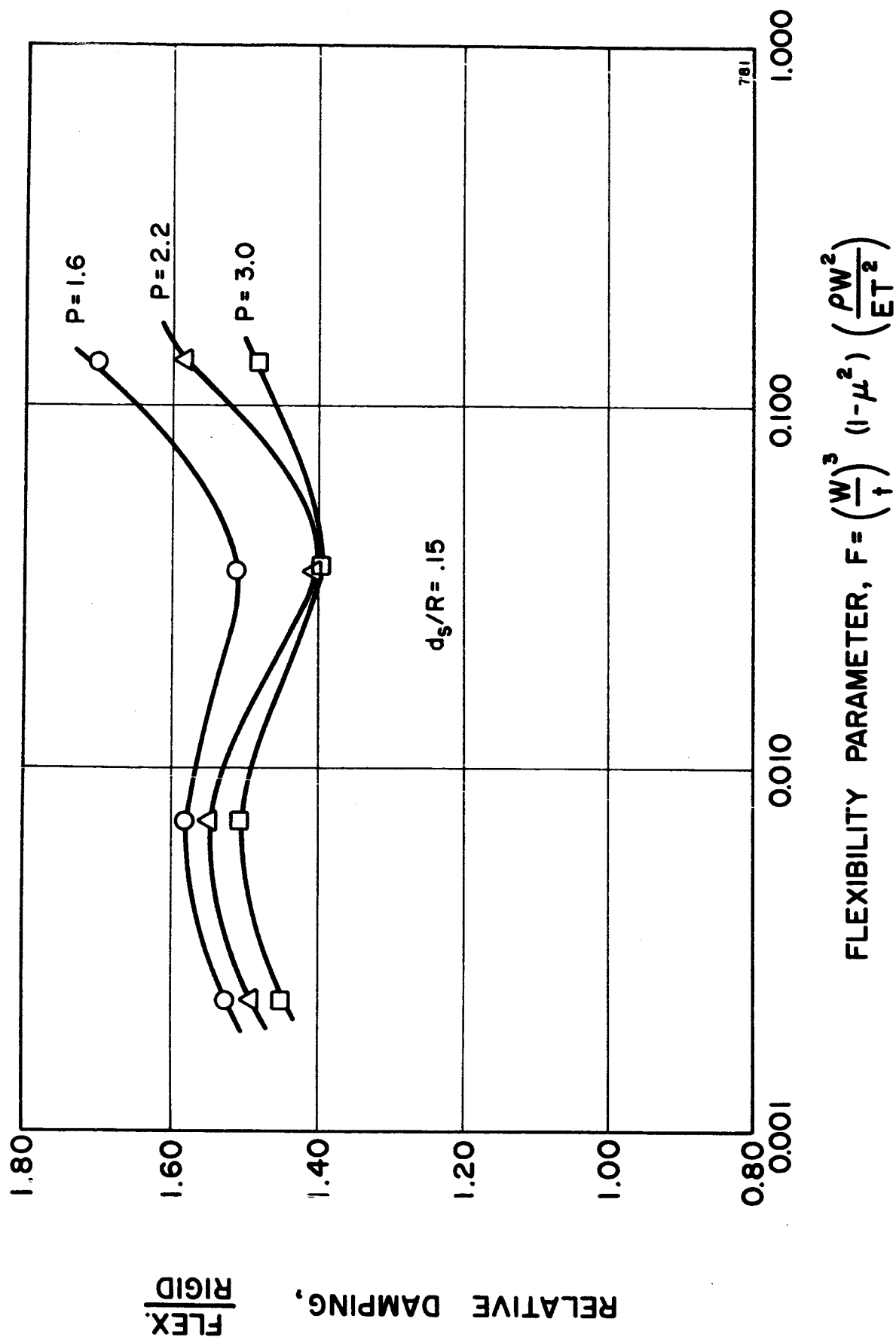


FIGURE 15. RELATIVE DAMPING vs FLEXIBILITY PARAMETER
FOR A BAFFLE DEPTH $d_g/R = 0.15$

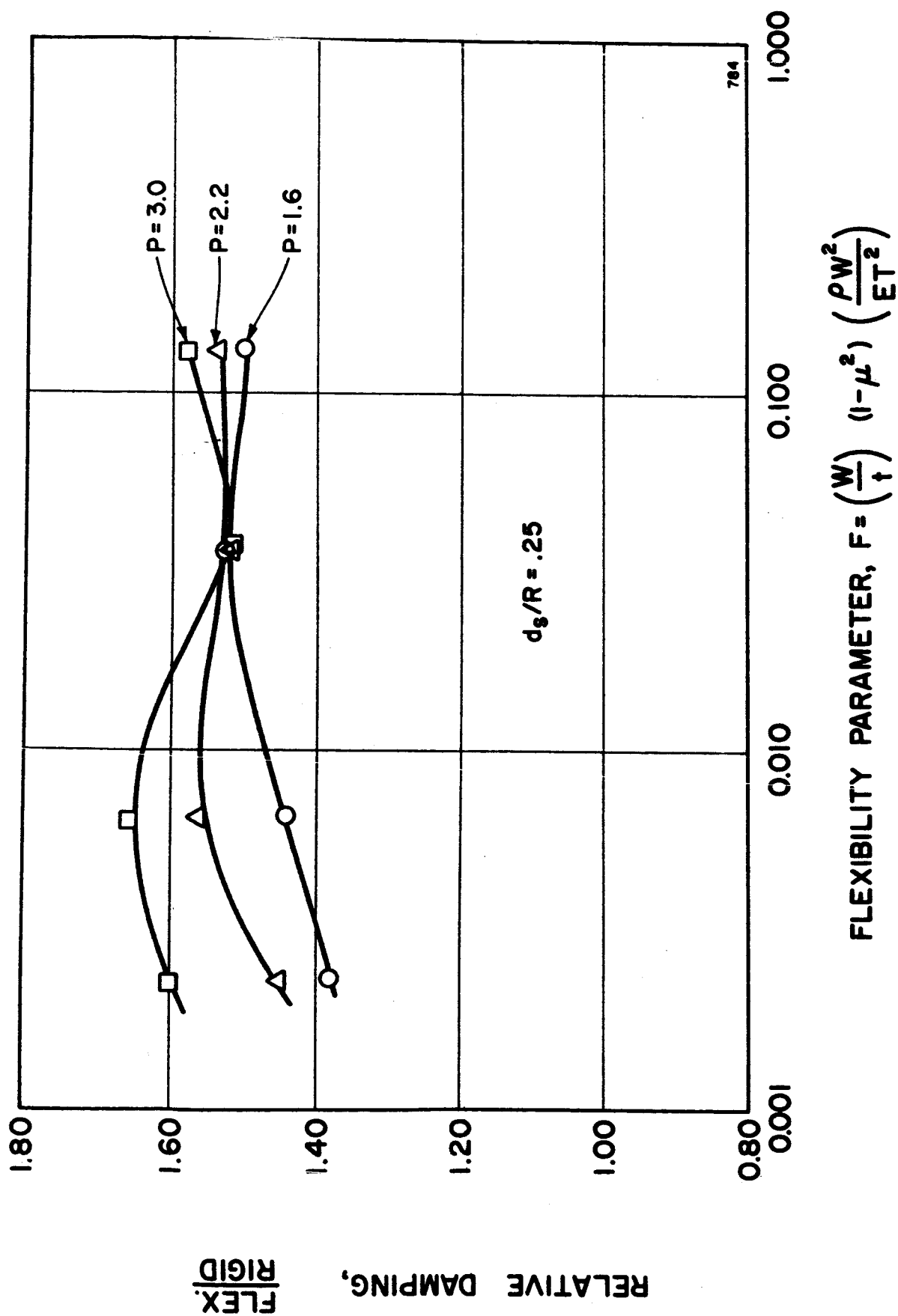


FIGURE 17. RELATIVE DAMPING vs FLEXIBILITY PARAMETER
FOR A BAFFLE DEPTH $d_g/R = 0.25$

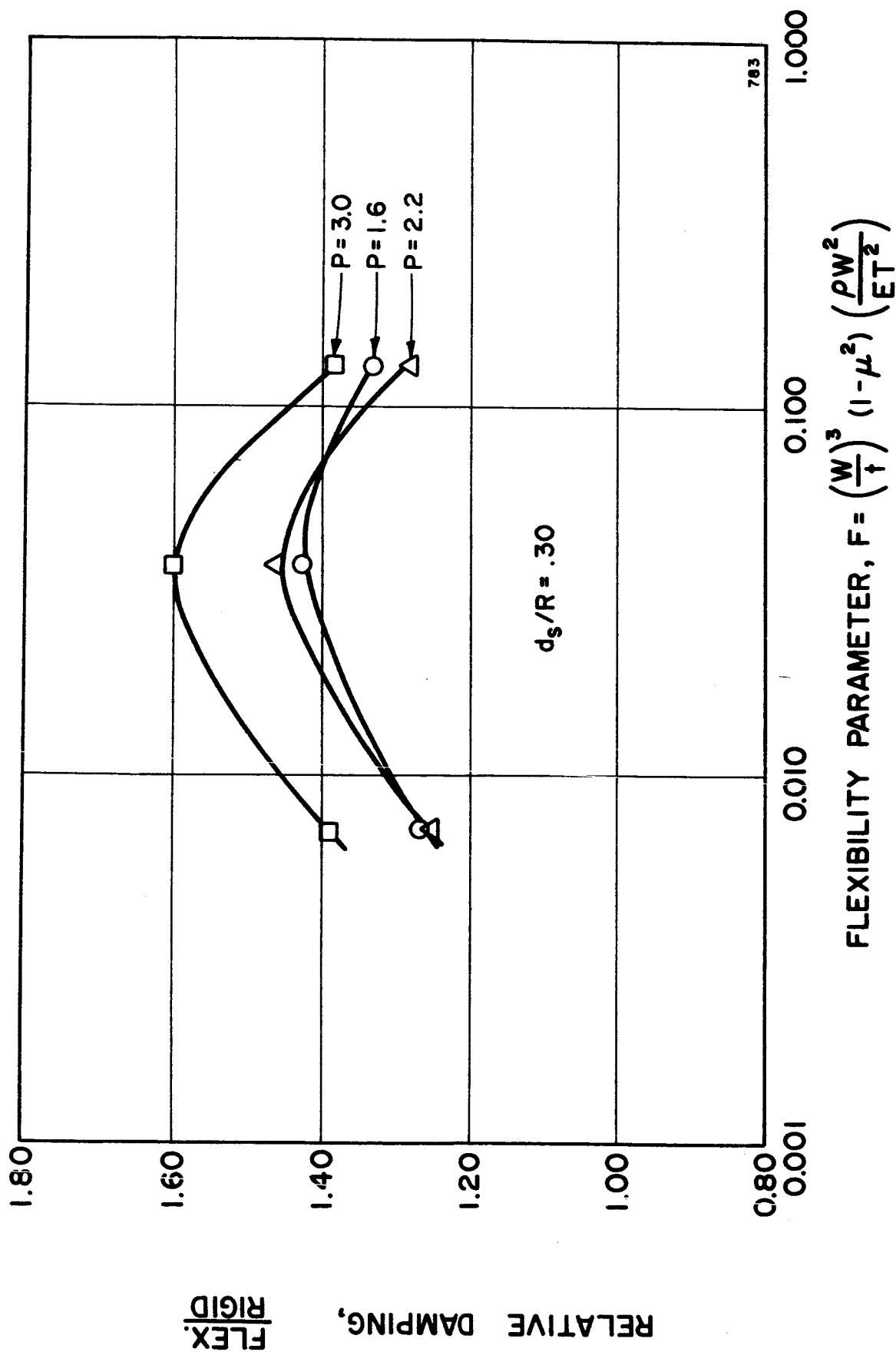


FIGURE 18. RELATIVE DAMPING vs FLEXIBILITY PARAMETER
FOR A BAFFLE DEPTH $d_s/R = 0.30$

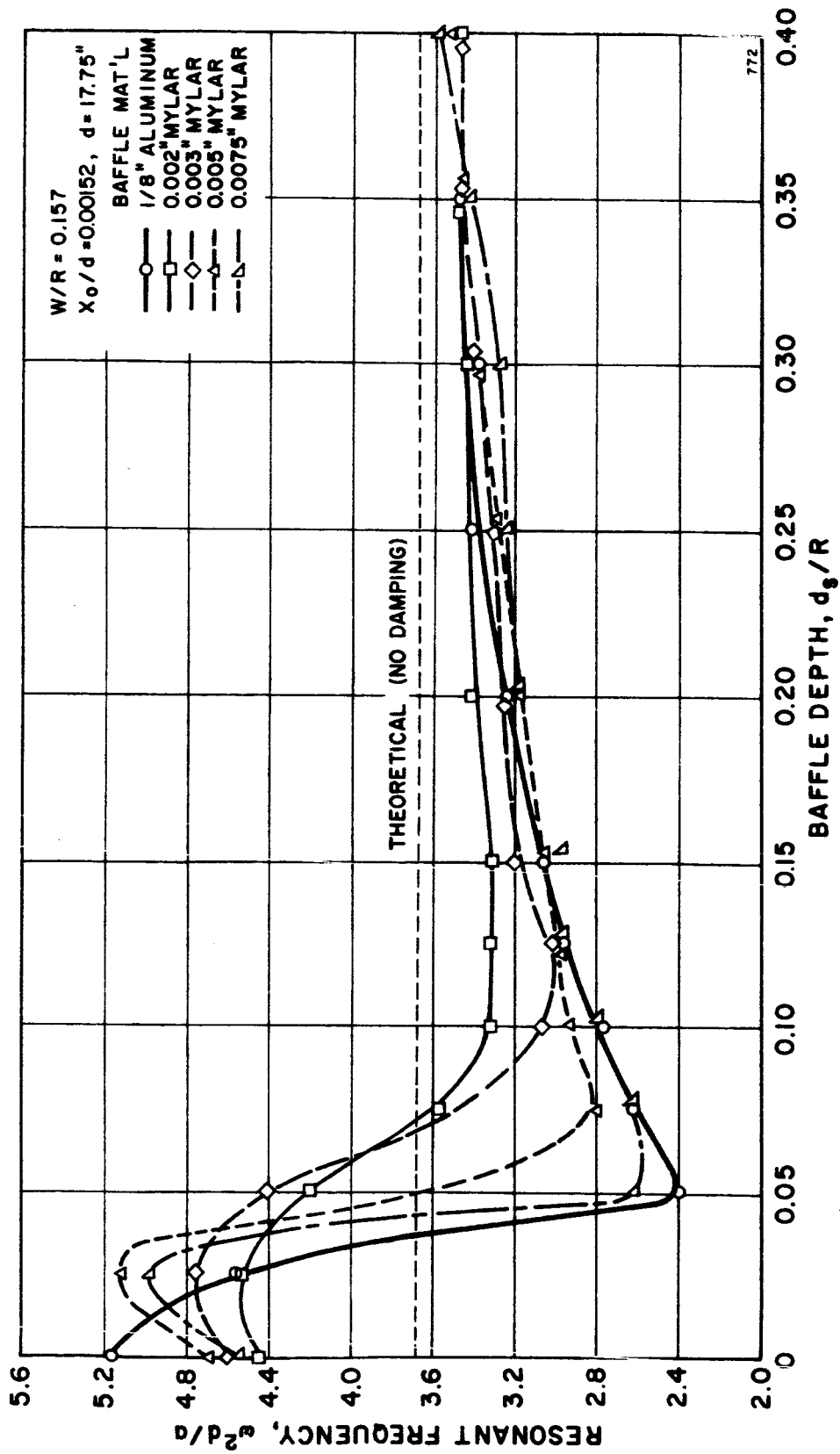


FIGURE 19. A COMPARISON OF THE EFFECT OF VARIOUS RING BAFFLES ON THE LIQUID RESONANT FREQUENCY FOR A TANK AMPLITUDE $X_0/d = 0.00152$

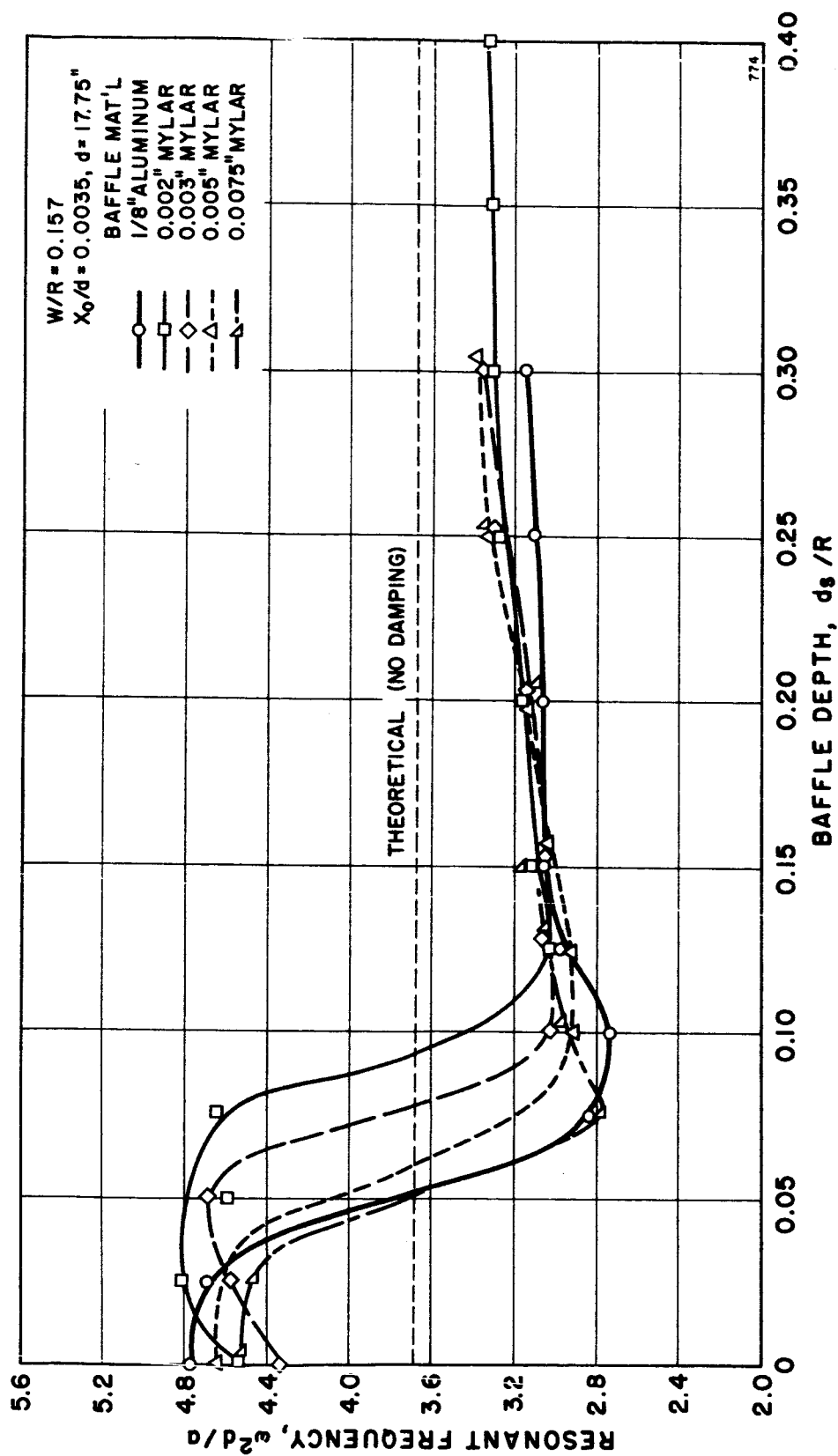


FIGURE 20. A COMPARISON OF THE EFFECT OF VARIOUS RING BAFFLES ON THE LIQUID RESONANT FREQUENCY FOR A TANK AMPLITUDE $X_0/d = 0.0035$

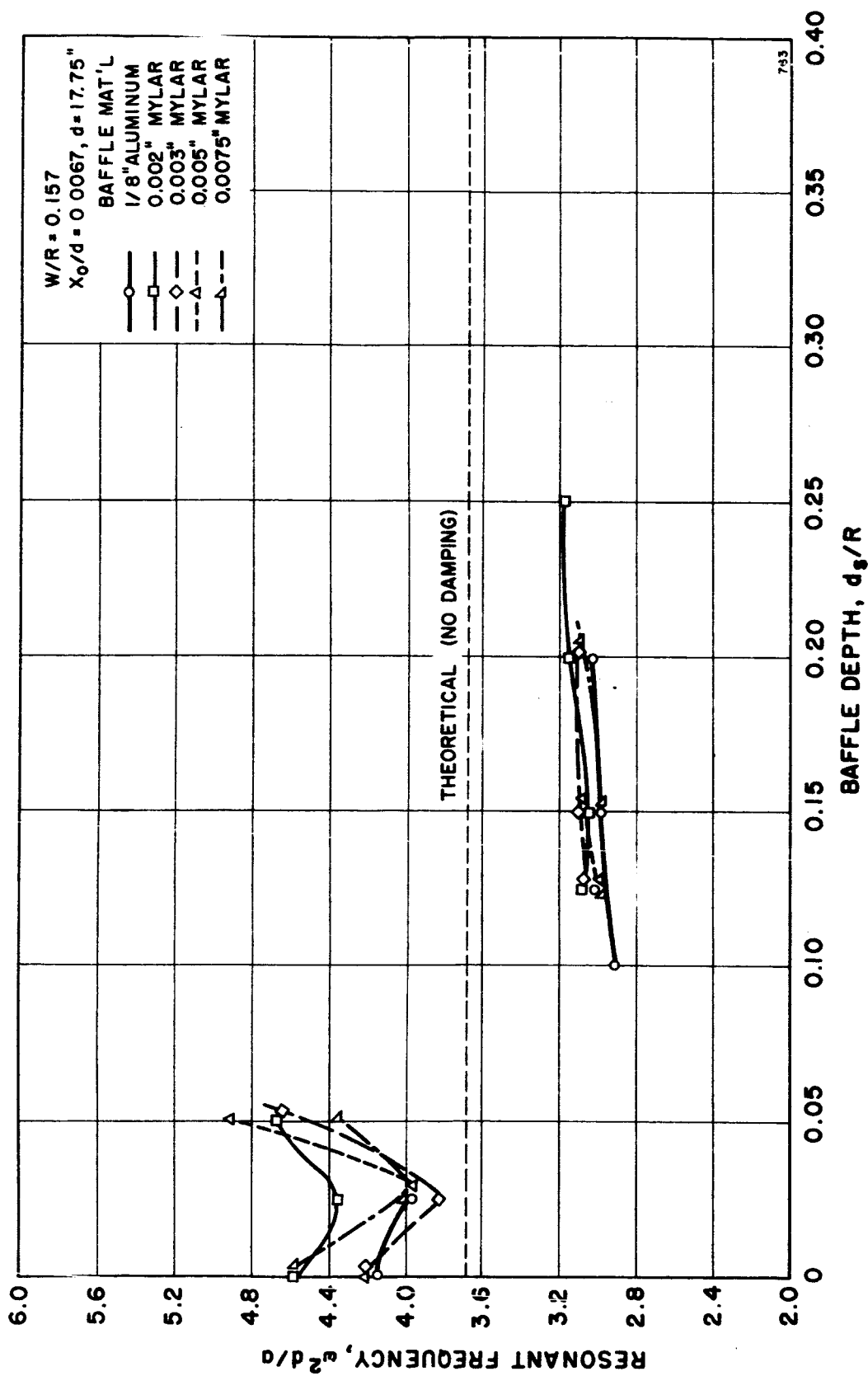


FIGURE 21. A COMPARISON OF THE EFFECT OF VARIOUS RING BAFFLES ON THE LIQUID RESONANT FREQUENCY FOR A TANK AMPLITUDE $X_0/d = 0.0067$

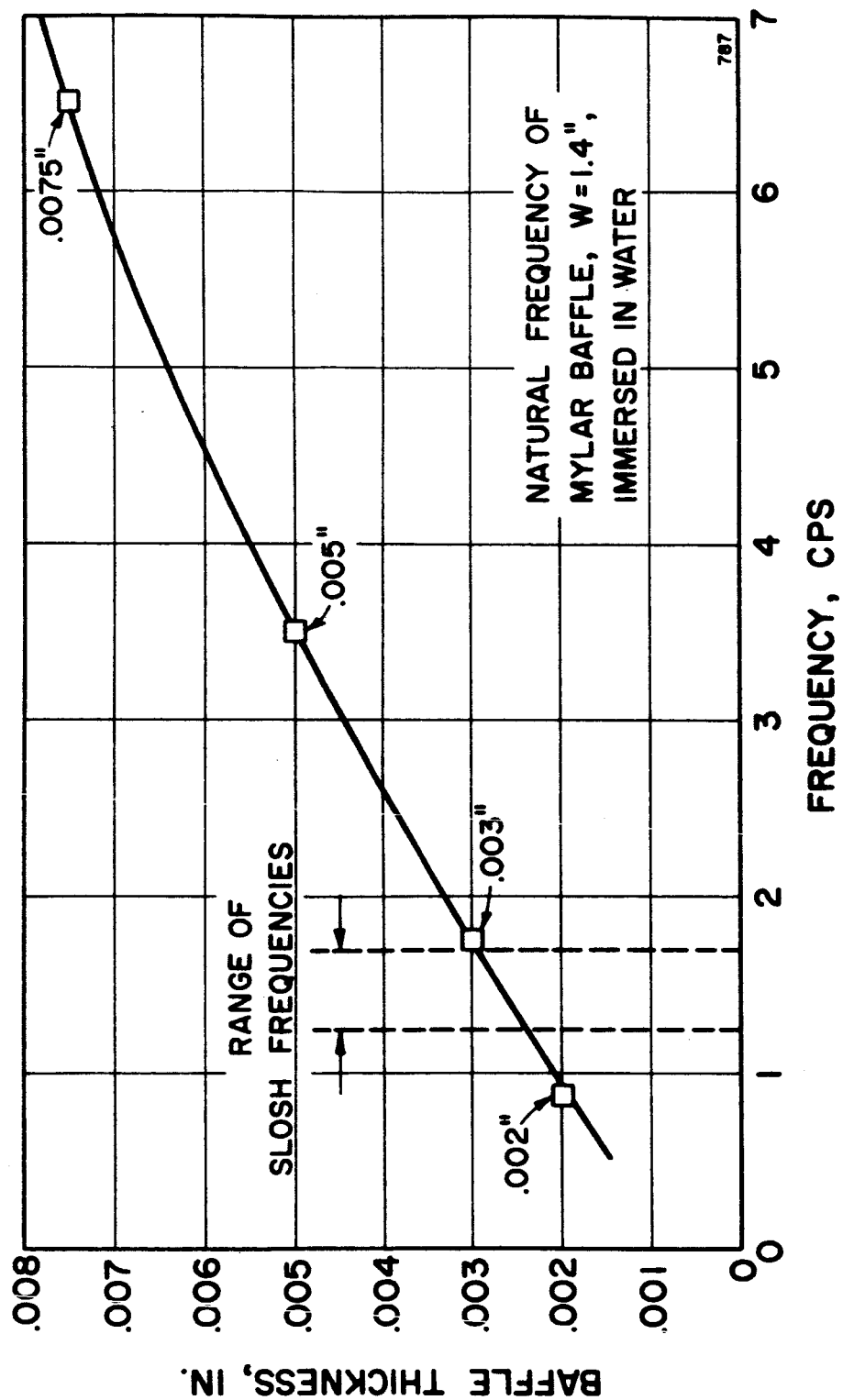


FIGURE 22. BAFFLE NATURAL FREQUENCY AS A FUNCTION OF BAFFLE THICKNESS